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Stuart, C., Harley, P., Spence, S., Kim, S., Filsinger, D., & Starke, A. (2014). *An analysis of stage performance in automotive turbocharger centrifugal compressors*. 1-10. Paper presented at Turbocharger Seminar 2014, Tianjin, China.

Document Version:
Peer reviewed version

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AN ANALYSIS OF STAGE PERFORMANCE IN AUTOMOTIVE TURBOCHARGER CENTRIFUGAL COMPRESSORS

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Abstract

As the concept of engine downsizing becomes ever more integrated into automotive powertrain development strategies, so too does the pressure on turbocharger manufacturers to deliver improvements in map width and a reduction in the mass flow rate at which compressor surge occurs. A consequence of this development is the increasing importance of recirculating flows, both in the impeller inlet and outlet domains, on stage performance.

The current study seeks to evaluate the impact of the inclusion of impeller inlet recirculation on a meanline centrifugal compressor design tool. Using a combination of extensive test data, single passage CFD predictions, and 1-D analysis it is demonstrated how the addition of inlet recirculation modelling impacts upon stage performance close to the surge line. It is also demonstrated that, in its current configuration, the accuracy of the 1-D model prediction diminishes significantly with increasing blade tip speed.

Having ascertained that the existing model requires further work, an evaluation of the vaneless diffuser modelling method currently employed within the existing 1-D model is undertaken. The comparison of the predicted static pressure recovery coefficient with test data demonstrated the inherent inadequacies in the resulting prediction, in terms of both magnitude and variation with flow rate. A simplified alternative method based on an equivalent friction coefficient is also presented that, with further development, could provide a significantly improved stage performance prediction.

Nomenclature

A	Flow area (m ²)
AR	Area ratio (-)
b	Passage height (m)
B	Blockage (-)
C_f	Equivalent skin friction coefficient (-)
CP	Static pressure recovery coefficient (-)
C_p	Specific heat at constant pressure (J/kgK)
D	Diameter (m)
\dot{m}	Mass flow rate (kg/s)
p	Static pressure (Pa)
p_0	Total pressure (Pa)
R	Radius (m)

U	Blade speed (m/s)
V	Absolute velocity (m/s)
W	Relative velocity (m/s)
π	Total-total pressure ratio (-)
γ	Ratio of specific heats (-)
α_2	Impeller tip flow angle relative to radial (°)
ϕ_{01}	Stage flow coefficient (-)
η	Isentropic total-total efficiency (-)
ρ	Density (kg/m ³)
β	Flow angle relative to meridional (deg)
β_{bl}	Blade angle (deg)
CORA	Compressor Off-Design Radial Analysis
1-D	One-dimensional
SFM	Swirl flow meter
VLD	Vaneless diffuser

Subscripts:

b	Blade
crit	Critical
ini	Initial
r	Radial direction
u	Tangential direction
1	Stage inlet
2	Impeller exit
3	Vaneless diffuser exit
4	Stage exit

1. Introduction

Increasing pressure on automotive manufacturers to deliver vehicles with lower fuel consumption figures and reduced emissions is placing engine boosting technologies at the forefront of vehicle powertrain development strategies. It is specifically the off-design conditions where the focus is currently being placed. The typical urban drive cycle mostly consists of engine part load running [1], which results in the turbocharger compressor operating away from the best efficiency point. In addition, the move towards engine downsizing to deliver reduced fuel consumption figures and CO₂ emissions, coupled with ever more aggressive use of exhaust gas recirculation (EGR) to control NO_x emissions [2], is placing increasing pressure on turbocharger manufacturers to improve compressor performance on the surge side of the performance map.

However, while 1-D techniques have a proven ability to accurately predict performance near the best efficiency

point, it is off-design conditions which are of the greatest interest for turbocharger applications and where further improvement in the existing modelling methods is required. The combination of these factors renders existing meanline techniques incapable of performance prediction with the required accuracy for the needs of modern automotive turbocharging applications.

Meanline modelling of impeller recirculation flows has proven to be a key element in improving 1-D performance predictions, having been shown to have a profound impact on performance close to the surge line [3]. With backswept blading at impeller exit being effectively ubiquitous in turbocharging applications, along with the associated increase in impeller inlet to outlet radius ratio, the presence of recirculating flows is a key feature in modelling the performance of turbocharger centrifugal compressors.

The current study presents an evaluation of the inclusion of the impeller inlet recirculation model of Harley *et al.* [4] on the 1-D performance prediction for two automotive turbocharger centrifugal compressors of substantially differing geometry (C-4 and C-5). In order to assess the improvement offered by the new modelling method, the 1-D results are compared against testing results gathered at QUB. As a means of providing guidance for future improvement of the existing 1-D model, an investigation into predicted performance of the vaneless diffuser based upon the static pressure recovery coefficient was also conducted. This was completed using the 1-D modelling results, as well as interstage static pressure data gathered during testing.

2. 1-D Centrifugal Compressor Modelling

The QUB Compressor Off-Design Radial Analysis (CORA) code has been developed over the past three years by Harley *et al.* [5]. It utilises a single-zone meanline approach and predominantly implements loss models available in open literature to provide a design and off-design performance prediction for turbocharger centrifugal compressors. An addition to the literature sourced loss models was the development of a correlation to describe the impact of recirculating flow at impeller inlet when operating close to surge, and it is described in more detail in subsequent sections. Despite a thorough evaluation of the most appropriate impeller loss models to capture the performance of turbocharger compressors, as well as developing a new meanline inlet recirculation model, improvements in model accuracy can still be achieved through further work.

Work conducted to date on the CORA meanline code has highlighted the vaneless diffuser model as an area for improvement. Currently, vaneless diffuser performance is predicted within the CORA code using the Herbert [6] model, which was a development and correction of Stanitz' [7] pioneering work. In order to quantify this observation, as well as to provide direction for future work, a comparison of the diffuser static pressure recovery coefficient predicted by the 1-D model and that of the testing data were conducted across a range of operating conditions.

2.1. Impeller inlet recirculation

The recognition of inlet recirculation as a key element in determining the performance of turbocharger centrifugal compressors near the surge point is not a new development, however the successful modelling of the impact on stage performance is a comparatively recent addition. Qiu *et al.* [3] presented a meanline model for the evaluation of impeller recirculation flows, encompassing both the inlet and outlet domains, and this formed the basis upon which the work of Harley *et al.* [4] on inlet recirculation was completed. A typical meridional view of inlet recirculating flow is depicted in Figure 1.

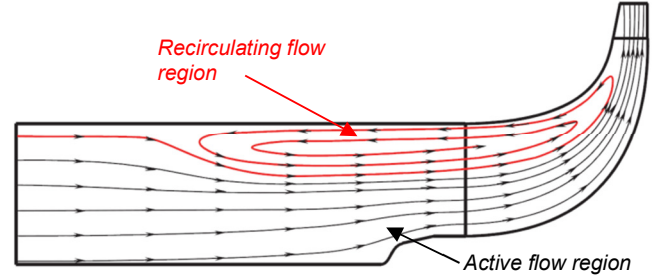


Figure 1: Meridional view of impeller inlet recirculation (adapted from [4])

Within the work of Qiu *et al.* [3], the extent of the recirculating flow region was defined in the spanwise direction by a “blockage” term, whereby this effectively isolated region of flow created an effective inlet diffuser from the inlet region to the throat. The area ratio, AR , of the inlet diffuser was assumed to be governed by the incidence angle (as depicted in Equation 1), whereby increasing the incidence angle increases the area ratio up until the point where the diffuser stalls.

$$AR = \frac{\cos(\beta_{1b})}{\cos(\beta_1)} \quad (1)$$

It is at this operating condition where the critical area ratio, AR_{crit} , is defined, which signifies the onset of recirculation and hence aerodynamic blockage within the flow passage. The analysis of how the extent of the blockage changed with mass flow rate was based upon this critical area ratio, which it was assumed throughout the analysis was equal to a value of 1.5. The implication of this value is that the recirculation fills a portion of the passage, while maintaining a ratio of 1.5 in the active flow region. It is therefore possible to solve for the inlet blockage value, B_1 , required to ensure the active flow region does not exceed the critical area ratio, as depicted in Equation 2.

$$B_1 = 1 - \frac{\dot{m}}{\rho_1 A_1 (U_1 - V_{u1})} \sqrt{\left(\frac{AR_{crit}}{\cos(\beta_{1b})} \right)^2 - 1} \quad (2)$$

Harley *et al.* [4] maintained the same method of calculating blockage, however the criterion for the onset of inlet recirculation was questioned. While Qiu *et al.* [3] utilised a

constant value, investigations on three automotive turbocharger compressors showed that AR_{crit} could be better represented by a quadratic function. The resulting best fit equation produced from scrutinising the results of single passage CFD simulations for each of the test compressors is depicted in Equation 3, and showed how the size of the inlet recirculation zone varies predictably with impeller geometry. This equation was derived upon having collapsed the data for each turbocharger onto a single graph by plotting the critical area ratio against stage flow coefficient, ϕ_{01} , and was found to be valid for blockage values greater than zero.

$$AR_{crit} = 160\phi_{01}^2 - 25\phi_{01} + 2.2 \quad (3)$$

(for $B_1 > 0$)

While Harley *et al.* [4] showed the improvement in the prediction of blockage for the new correlation, the impact on the overall performance prediction of the CORA model was not evaluated. In order to quantify the impact of the inclusion of the inlet recirculation model developed by Harley *et al.* [4] on predicted compressor performance, the current study presents a comparison of the predicted performance maps generated by the QUB CORA code with and without the aforementioned recirculation model. The comparison was conducted for two automotive turbocharger centrifugal compressor stages of substantially differing geometry, and the overall improvement in the modelling approach was evaluated using testing data gathered within QUB.

2.2. Test Facility

The entirety of the testing data gathered to undertake this study was collected from the hot gas stand test rig in the QUB Flow Lab. The layout and key components of the test facility are illustrated in Figure 2.

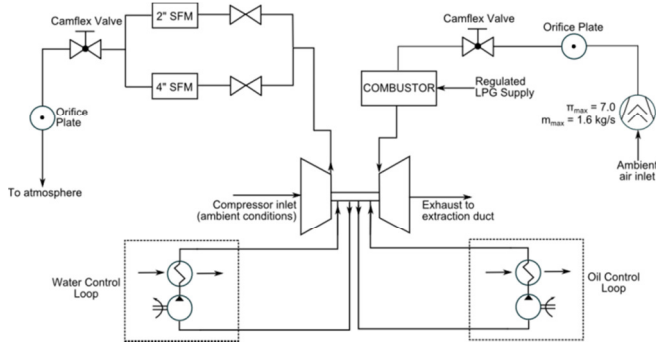


Figure 2: QUB Turbocharger test facility schematic

As is depicted in Figure 2, the basic layout of the test rig follows that described in SAE J1826 [8] for two-loop turbocharger hot gas test stands. On the compressor side, air is induced from ambient conditions, and is discharged into a pipeline which branches to offer the choice of a 2" or 4" swirl flow meter (SFM) depending on the compressor

operating mass flow rate. The possibility of utilising an orifice plate to measure compressor mass flow is also present, however having verified the accuracy of the swirl flow meters, the orifice plate has been kept for use as a backup. Compressor mass flow (and hence pressure ratio) is controlled by throttling the discharge line with the Camflex valve shown.

On the turbine side of the rig, compressed air is supplied by a variable speed screw compressor. Mass flow rate is measured using a range of orifice plates, and is controlled with another Camflex valve as shown in Figure 2. The air then passes to a combustor, where it is mixed with a pressure regulated liquid petroleum gas (LPG) supply and ignited to increase the turbine inlet temperature to the desired level. Having been expanded through the turbine, the spent gases are discharged into extraction ducting.

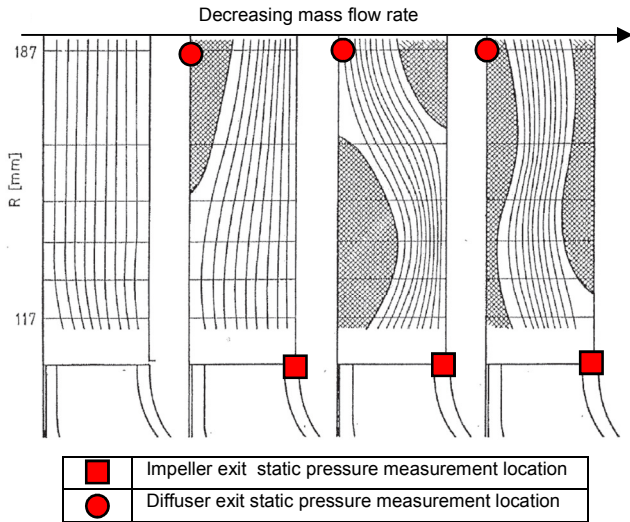
2.2.1. Instrumentation

Mass flow measurements were collected using one of the two ABB FS4000-ST4 swirl flow meters depicted in Figure 2, to an error of less than $\pm 0.5\%$ at reference flow conditions [9]. Static pressure data was gathered using Druck PMP 4000 Series gauge pressure transducers, delivering readings to an accuracy of $\pm 0.04\%$ full scale [10]. Temperature measurements on the compressor side of the rig, as well as at turbine discharge, were logged using PT-100 resistance thermometers. By comparison, turbine inlet temperatures were gathered using K-type thermocouples, which were necessary due to the possible maximum temperature being far in excess of the 200°C limit of the PT-100 sensors. During data post processing, in order to obtain a pressure or temperature value at a particular position, the numerous circumferential values were averaged in order to obtain a single value.

Extensive interstage static pressure data for each operating point was gathered using wall tapings at the inducer tip, impeller exit, vaneless diffuser exit and within the volute. Measurements were gathered at six circumferential locations at the inducer tip and impeller exit, twelve locations at vaneless diffuser exit and 36 locations within the volute. The position of these measurements was chosen to coincide with 1-D modelling interstage locations. The impeller exit pressure taps were located on the shroud wall, while the vaneless diffuser exit pressure taps were located on the hub wall, as illustrated in Figure 3. The choice of hub or shroud location for these pressure tapings was strictly governed by packaging issues, rendering it impractical to choose any other positions.

It is clear from examining Figure 3 that, depending on operating condition, the position of the pressure tapings could yield an incorrect perception of the actual static pressure due to the presence of separated flow, and the curvature of the mainstream flow. It is worth noting however, that while this will undoubtedly impact upon the exact values of the results obtained at low mass flow rates, it is unlikely that it will have a perceivable impact on the

overall trends observed. Further analysis was nonetheless warranted.



**Figure 3: Vaneless diffuser separated flows
(Adapted from Rebernick [11])**

In order to help quantify the error brought about by the inability to have pressure tapings on both the hub and shroud walls, an investigation was carried out using a single passage CFD simulation for C-4. The setup of the model mirrors exactly what was used in previous work [4]. ANSYS-CFX was once again the chosen CFD tool, with the “Turbo Line” feature being used to plot the variation of static pressure from hub to shroud at impeller exit. The study could not be extended to evaluate the variation at vaneless diffuser exit due to the inability to account for the volute within a single passage simulation.

It was determined that across all of the operating conditions investigated (encompassing low, medium and high tip speeds at surge, choke and mid-map flow conditions), the maximum difference between the hub and shroud static pressure (on an absolute basis) was 2.0%. From this result it is clear that the variation in static pressure from hub to shroud will not have a significant impact upon the CP results calculated from testing data.

2.2.2. Heat transfer

A key element impacting the apparent efficiency of the compressor stage is the effect of heat transfer. Heat transfer in turbochargers can be broadly categorised as internal or external. External heat transfer, whereby heat is lost from the compressor to the surroundings, can be effectively reduced by insulating the compressor housing. This was addressed on the test rig by wrapping the turbocharger in fiberglass mat before the commencement of the testing program. By comparison, in order to minimise the impact of internal heat transfer on test results, the only truly accurate method is to undergo a process of thermal matching at each operating point, whereby the compressor discharge, oil inlet and turbine inlet temperatures are equalised [12]. While in the absence of inlet temperature regulation compressor

discharge temperature cannot be controlled (it is purely a function of operating conditions), both the turbine inlet temperature and bearing housing temperature can be controlled on the test rig. As illustrated in Figure 2, control loops for the temperature of both the lubricating oil and coolant (where applicable) are present on the test rig. This allows PID control of these parameters via cooling fans, thus ensuring that the desired operating temperatures are maintained within the bearing housing.

Due to limitations in ensuring that formation of ice or condensation did not occur at turbine outlet during testing, this criterion could not be met in its entirety. However, throughout the testing procedure it was possible to maintain a difference of no more than 30°C between the turbine inlet temperature and compressor discharge temperature for each operating point, thus minimising the impact of internal heat transfer on compressor isentropic efficiency.

In order to quantify the degree of heat transfer present within the testing results, the data was corrected using the procedure presented by Sirakov and Casey [13]. This method allows data gathered under diabatic test conditions to be converted to performance maps for adiabatic conditions, without the need for direct measurement of heat transfer. The correction procedure was completed for both C-4 and C-5, the results of which are presented in Figure 4 and Figure 5.

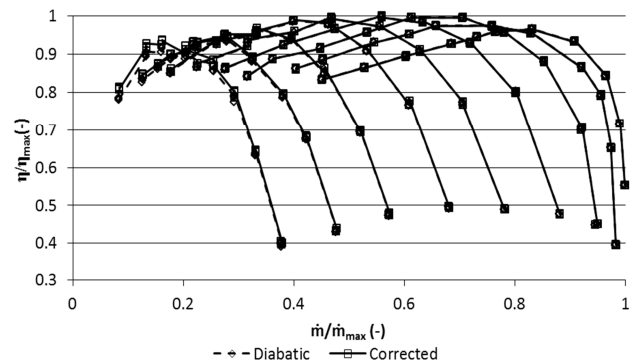


Figure 4: Comparison of diabatic and corrected efficiency test data for C-4

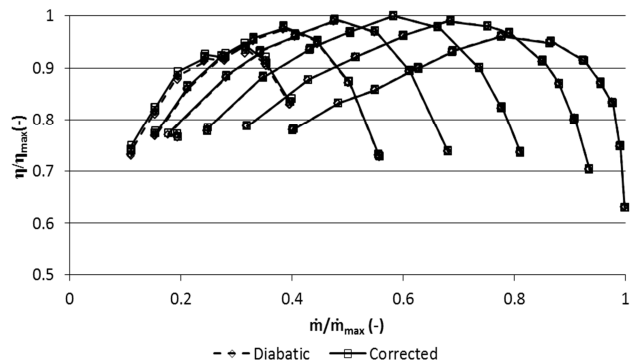


Figure 5: Comparison of diabatic and corrected efficiency test data for C-5

Upon studying the results depicted in Figure 4 and Figure 5, it is readily apparent that the procedure undertaken during testing of the compressors to minimise the impact of heat transfer on the results has been very successful. For both C-4 and C-5 it is clear that there is no perceivable impact of heat transfer on the isentropic efficiency results. With traditional hot gas stand testing utilising turbine inlet temperatures representative of those encountered during on-engine operation, it would not be unreasonable to expect an under prediction of 20% in compressor efficiency at low speeds [13]. With the current data set however, it is clear that the care taken in minimising the temperature difference across the turbocharger during testing has made it possible to neglect the effects of heat transfer without a significant impact on accuracy of the data.

3. Results and Discussion

3.1. Inlet recirculation model

From the results of the performance characteristic comparison presented in Figure 6 to Figure 9, it is clear that the incorporation of the inlet recirculation model has resulted in an improvement in the 1-D prediction of both pressure ratio and efficiency at low mass flow rates when compared to the testing data. Looking first at the pressure ratio characteristic for both compressors, it is clear that the inclusion of inlet recirculation within the modelling method has allowed the 1-D model to capture the characteristic flattening of the pressure ratio curve witnessed with turbocharger compressors as the mass flow rate is reduced. Upon examining Figure 6 and Figure 7 it is clear that without inlet recirculation, the model fails to capture this fundamental characteristic present in both sets of test data.

However, what is also readily apparent is that pressure ratio is over predicted by an ever increasing amount for both C-4 and C-5 as compressor speed increases. Taking the surge point from the test data at 100% speed as an example for both C-4 and C-5, it is clear that pressure ratio is over predicted by the 1-D model by 22.2% and 7.67% respectively. Furthermore, for the case of C-5, it is clear that the discrepancy at the surge point is not indicative of the error witnessed across the entire map, with a difference of 22.5% evident close to the peak efficiency point. While this is a substantial improvement over the values of 33.7% and 28.2% witnessed without the inlet recirculation model, it is clear that the pressure ratio over prediction is endemic across the entire width of the map. In addition to this, the prediction of the surge and choke points can be seen to deteriorate as speed increases.

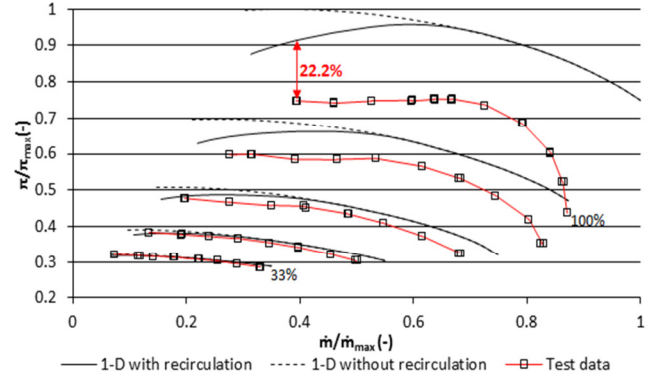


Figure 6: Comparison of pressure ratio characteristic between 1-D model with and without inlet recirculation, and testing results for C-4

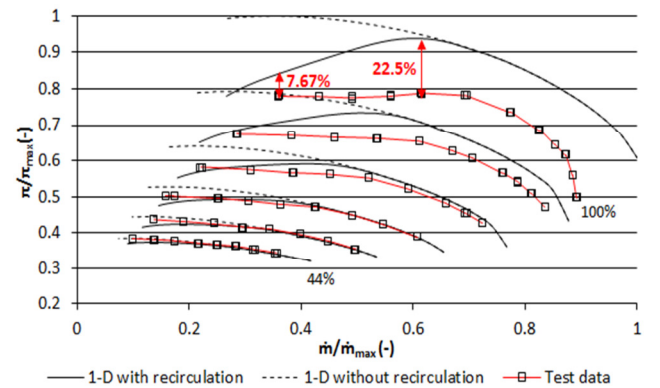


Figure 7: Comparison of pressure ratio characteristic between 1-D model with and without inlet recirculation, and testing results for C-5

Having examined the findings from the pressure ratio comparisons, analysis of the efficiency characteristics also yields some important findings. Figure 8 and Figure 9 again illustrate the considerable improvement in correlation with the test data brought about by inclusion of the inlet recirculation model, with good matching being achieved at low speeds for both compressors.

However, as was the case with the pressure ratio, increasing rotational speed leads to a divergence in the efficiency correlation between the 1-D predictions and the test data, coupled with an increasingly poor prediction of the choking mass flow rate. Taking all of the above findings into consideration, this gives clear evidence for the need for further work to improve modelling accuracy, a portion of which is presented in the following sections.

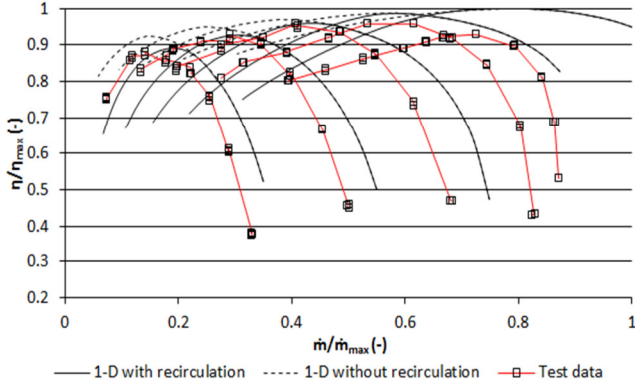


Figure 8: Comparison of efficiency characteristic between 1-D model with and without inlet recirculation, and testing results for C-4

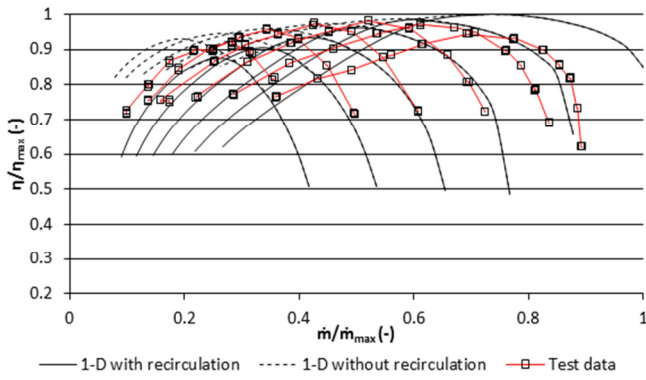


Figure 9: Comparison of efficiency characteristic between 1-D model with and without inlet recirculation, and testing results for C-5

3.2. Vaneless Diffuser performance investigation

The importance of impeller inlet recirculation flow on stage performance near the surge side of the map has been demonstrated, and the improvement brought about by the inclusion of recirculation modelling to an existing 1-D modelling tool has been quantified. However, comparing the 1-D modelling predictions with test data has shown that further work is required to improve the pressure ratio prediction. Work to date has shown that the vaneless diffuser model is an area for significant improvement, the extent of which will be investigated here.

In order to evaluate the ability of the Herbert vaneless diffuser model [6] to capture actual diffuser performance, an analysis of diffuser static pressure recovery was undertaken for C-4 and C-5. The static pressure recovery coefficient, CP, was defined in each case as the proportion of dynamic pressure at impeller exit that was successfully converted to static pressure at diffuser exit, as depicted in Equation 4.

$$CP = \frac{p_3 - p_2}{p_{02} - p_2} \quad (4)$$

The parameters required to undertake the analysis of CP for the 1-D model were directly extracted from the CORA code. By comparison, the unavailability of impeller exit total pressure from the testing results required some manipulation of the data to permit calculation of CP values for each operating point. The procedure undertaken is outlined in Figure 10 in order to provide clarity in terms of the method employed, depicting the input values for the analysis available from testing data as well as the key steps involved in the calculation procedure. The key assumptions applied within the method were, firstly, constant total temperature from impeller exit to stage exit and, secondly, an absence of inlet swirl thus allowing the Euler Turbomachinery Equation to be used to determine the flow absolute tangential velocity at impeller exit, V_{u2} .

While the use of such simplifying assumptions will not exactly replicate reality, steps have been taken to ensure the validity of the results obtained. As highlighted in Section 2.2.2, it has been shown that heat transfer effects within the testing data are negligible for both compressors, thus providing some verification for the assumption of constant total temperature from impeller to stage exit. Furthermore, in order to ensure the validity of the assumption of zero inlet swirl, the inlet section to both compressors was modified prior to testing. The modifications undertaken removed any non-uniform geometry immediately upstream of the impeller that could introduce secondary flow patterns, and hence negated the likelihood of any associated non uniformity of the inlet flow due to geometric variations.

In order to provide validation of the method employed to calculate CP from the test data, the results for both C-4 and C-5 were compared with the CP data presented in the

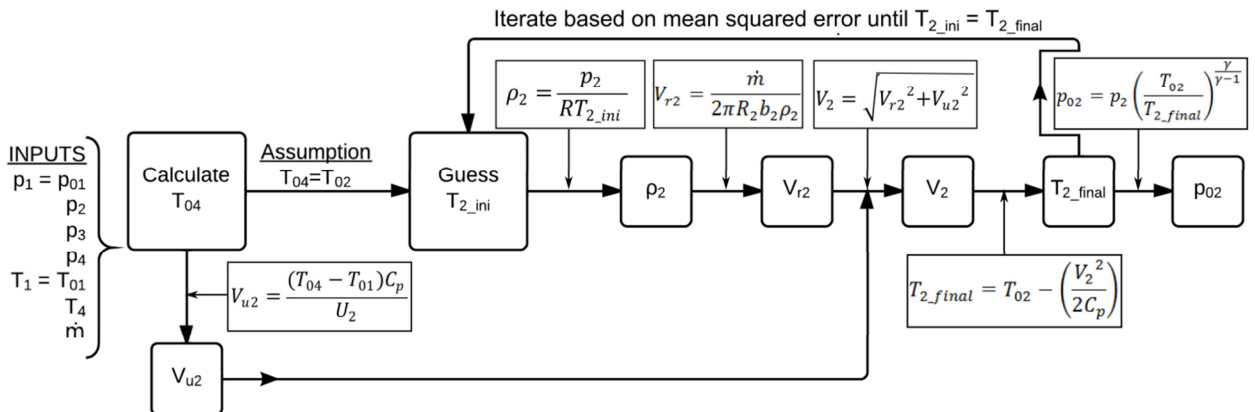


Figure 10: Calculation procedure for impeller exit total pressure

literature. The experimental data depicted in Figure 11 was presented by Rodgers [19], and represents the best fit line for eleven different diffusers of similar geometry to those tested in the current study. It is readily apparent from the comparison that the test data of both C-4 and C-5 replicates the positive correlation between diffuser CP and the ratio of tangential to radial absolute velocity at diffuser inlet. The trend witnessed can be explained by flow path length; as the value of V_{R2}/V_{U2} decreases at lower mass flow rates, the flow becomes more tangential, giving rise to a longer flow path within the diffuser and hence increased frictional losses. This then manifests itself in a reduced value of CP.

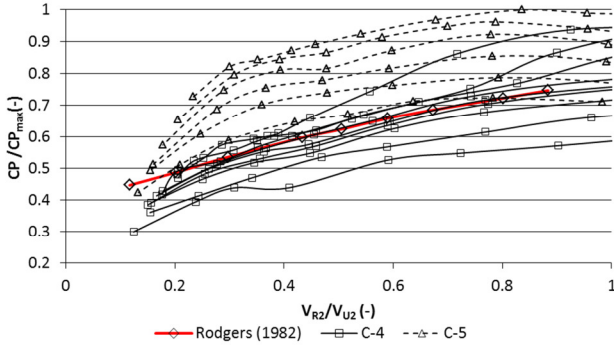


Figure 11: Comparison of C-4 and C-5 tested CP values with those of Rodgers [19]

Having undertaken the analysis for the same two automotive turbocharger applications as detailed in the previous section, the results are depicted in Figure 12 and Figure 13 for C-4 and C-5 respectively for both the test data and the 1-D prediction. What is immediately striking about the results is that while both methods predict increasing CP with increasing mass flow rate for both compressors, the 1-D prediction does not capture the magnitude of variation of CP across the operating range. This clearly highlights a significant source of error within the current 1-D model.

It is interesting to note that for both of the tested compressors, and the data presented by Rodgers [19], that the diffuser is unstable across almost the entirety of its operating range. As shown by Abdelhamid [14], a pressure rise which increases with flow rate is always unstable. It is only C-5 which exhibits any significant region of diffuser stability towards the upper end of mass flow rate for a given speed line, as signified by the negative slope of the tested pressure rise in Figure 13. It is clear therefore that the vaneless diffuser is being kept from stall and eventual surge across the majority of the map by the impeller pressure rise characteristic sloping in the opposite direction.

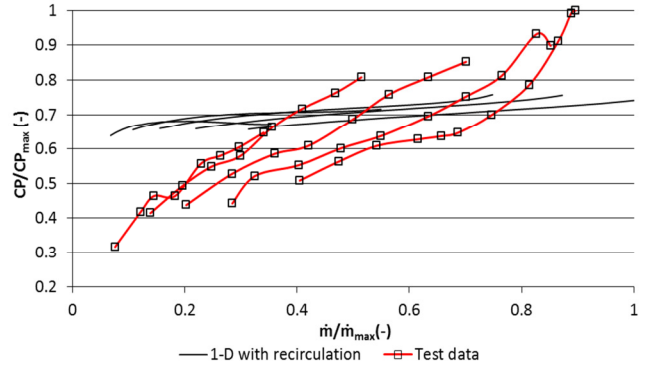


Figure 12: Comparison of vaneless diffuser static pressure recovery coefficient predicted by 1-D modelling and test data for C-4

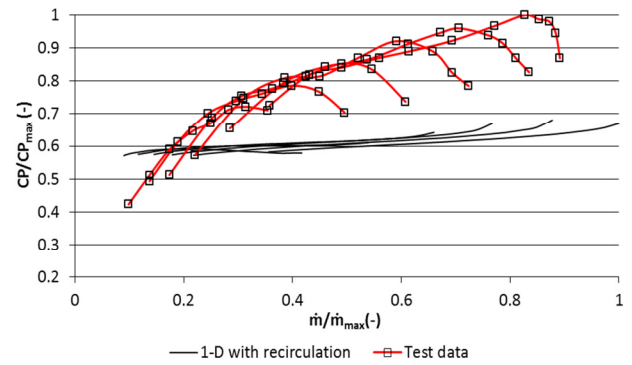


Figure 13: Comparison of vaneless diffuser static pressure recovery coefficient predicted by 1-D modelling and test data for C-5

As a result of the above comparisons for both C-4 and C-5, it is readily apparent that while the predicted positive gradient is correct, the Herbert vaneless diffuser model is not correctly predicting the performance of the vaneless diffuser at the extremes of the compressor operating range.

3.3. Improving the existing 1-D prediction

In order to locate the source of the error in the current vaneless diffuser performance prediction, the decision was taken to break down the CP calculation into constituent parts. This would permit analysis of how the individual terms making up the calculation vary with flow rate, and hence make it possible to pin point specific weaknesses in the method. Having undertaken this process for both C-4 and C-5, it was immediately apparent that in fact all elements of the calculation have a contributing role in the inaccuracy of the prediction.

Taking C-5 as an example and looking first at the normalized static pressure rise across the diffuser (where the normalizing term was the maximum static pressure value) as depicted in Figure 14, it is clear that the Herbert model not only under predicts the magnitude, but also shows an incorrect variation of static pressure rise with mass flow rate. While the testing data shows a comparatively constant

static pressure rise with flow rate for any given speed line, the 1-D prediction delivers an ever increasing variation as operating speed increases.

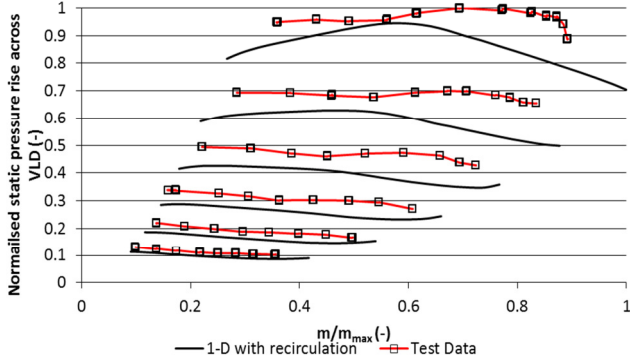


Figure 14: Normalised static pressure rise across vaneless diffuser of C-5

A similar magnitude of discrepancy is witnessed for the dynamic pressure available at impeller exit / vaneless diffuser inlet, as illustrated in Figure 15. Once again, the trend depicted by the test data in terms of variation with mass flow rate is not correctly captured by the 1-D model, showing that, as previously stated, all elements of the calculation of CP have a contributing role to play in the resulting inaccuracy.

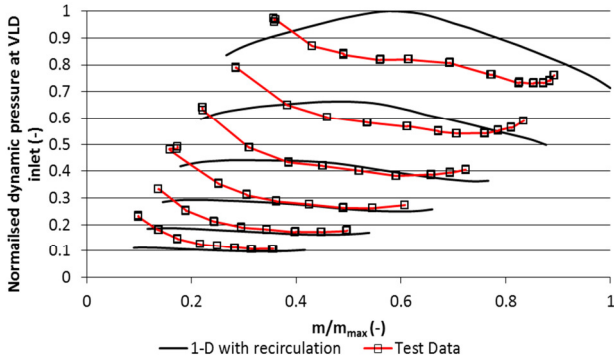


Figure 15: Normalised dynamic pressure at vaneless diffuser inlet of C-5

The discrepancy in dynamic pressure available could be attributed to not accounting for mixing of the characteristic jet / wake flow pattern emanating from the impeller. The previous work by Harley *et al.* [5] utilised the impeller loss model collection of Galvas [15], which does not account for exit mixing, as most representative of the automotive turbocharger compressor stages being tested. This is a critical consideration in single zone modelling, where jet and wake flows cannot coexist either within impeller passages or within the diffuser. A further development of the current model to account for exit mixing in a similar method to Johnston and Dean [16], whereby a sudden expansion mixing loss was applied to the impeller exit flow to permit a steady, 1-D treatment of the flow throughout the diffuser, coupled with the associated decrement in total pressure

delivered by the application of such a loss, would significantly improve the modelling prediction.

Alternatively, manipulation of the skin friction coefficient in the diffuser entry region to account for the increased losses present in this section could also deliver a similar result. It is stated by Rodgers [19] and Bammert [17] that the vaneless diffuser can be well predicted, even with very distorted inlet flows, using friction coefficient alone. It was noted that a higher value in the region up to a radius ratio of approximately 1.15 to account for mixing would be required, followed by a constant value for the rest of the diffuser. This was provided the boundary layers did not merge, which it was said was unlikely for vaneless diffuser diameter ratios of less than 1.8. This matches the findings of Dubitsky and Japikse [18], whereby it was noted that fine-tuning of the friction coefficient in the entry region can result in a good global diffuser performance prediction, but will not correctly predict the variation in flow parameters across the diffuser path. However it could be argued that, for a 1-D model, it is only the global predictions that are of importance.

Having analysed the data and consulted with the literature, it was decided to step back from the comparatively complex methodology applied by Herbert [6] and take a first principles approach to predicting vaneless diffuser performance. Taking into account the importance of the friction coefficient on diffuser performance, the method of Rodgers [19] for calculating an equivalent skin friction factor, C_f , was employed on the test data, as shown in Equation 5. This is effectively an average value representative of the entire diffuser, and could be further subdivided to permit evaluation of the impact of losses other than wall friction.

$$C_f = \frac{\left[\left(1 - \left(\frac{D_2}{D_3} \right)^2 \right) - CP \right] \cos(\alpha_2)}{\frac{D_2}{b_2} \left(1 - \frac{D_2}{D_3} \right)} \quad (5)$$

The method employed to implement this approach within the current study was to use Equation 5 to extract representative effective friction coefficient from test data for the entire diffuser as validation of the method for the current data set. It is worth noting that additional considerations including the influence of blockage and Reynolds number effects on the variation of C_f were not accounted for in this analysis; a proof of concept was the only aim at this stage. Once this representative value was extracted for both test cases, Equation 5 was rearranged to calculate CP from 1-D model. This involved the use of the representative friction coefficient extracted from the test data, stage geometry and impeller exit flow angle from the 1-D model. The resulting calculation employed is illustrated by Equation 6.

$$CP = \left[1 - \left(\frac{D_2}{D_3} \right)^2 \right] - \frac{C_f}{\cos(\alpha_2)} \frac{D_2}{b_2} \left(1 - \frac{D_2}{D_3} \right) \quad (6)$$

This analysis was undertaken for both C-4 and C-5, the results of which are illustrated in Figure 16 and Figure 17 respectively.

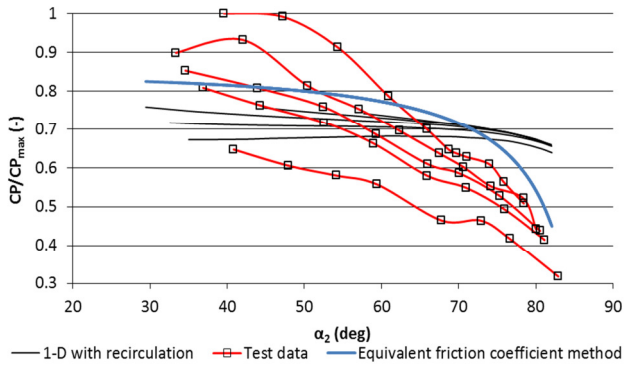


Figure 16: Variation of CP with flow angle for C-4

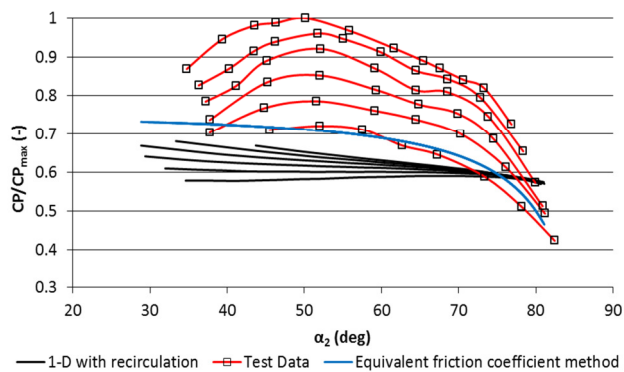


Figure 17: Variation of CP with flow angle for C-5

The results depicted in Figure 16 and Figure 17 clearly illustrate that, for both compressor cases, while the magnitude of the prediction of the equivalent friction coefficient method may vary slightly from the test data, it does in fact capture the correct variation with impeller exit flow angle and hence mass flow rate. It is clear therefore that, despite its comparative simplicity, the equivalent friction coefficient method has had greater success in predicting variation of CP than the Herbert [6] model, as employed within the QUB CORA code (represented by the “1-D with recirculation” data series).

It is worth emphasising again that the results presented do not represent the extent of the capability of the equivalent friction coefficient method. Further work in refining the method, thereby improving the correlation with the test data and permitting variations in CP with operating speed is currently in progress, and it is envisioned will deliver a substantially improved prediction over the current method.

4. Conclusions

The improvement in the 1-D performance prediction for automotive turbocharger applications from inclusion of the inlet recirculation model of Harley *et al.* [4] has been quantified for two test cases, demonstrating significant

improvements towards the surge side of the map. However, it has been shown that the QUB CORA model requires further refinements to the modelling approach to remove the tendency to deliver diminishing prediction accuracy as compressor rotational speed is increased.

Analysis work undertaken to date on improving the QUB CORA model has highlighted the existing vaneless diffuser model as an area for improvement. Upon comparing the predicted diffuser CP delivered by the Herbert model with that of test data, it became clear that the existing model was not adequately capturing the variation of diffuser CP with mass flow rate. Further investigation undertaken by evaluating each component of the CP calculation yielded the finding that the prediction of both static pressure rise across the diffuser, and dynamic pressure available at diffuser inlet were incorrect when compared to the test data, in terms of both magnitude and variation with flow rate. This brought into question the absence of an impeller exit mixing term in the 1-D model, as well as the skin friction coefficient calculation method employed.

Application of an equivalent friction coefficient method akin to that published by Rodgers [19] as an alternative to the Herbert model resulted in a substantial improvement in CP prediction, that with further development in the areas outlined, could deliver a much improved performance prediction.

Acknowledgements

The authors would like to thank IHI Charging Systems International GmbH for their technical support and provision of the required compressor geometry.

References

1. Barlow, T.J., Latham, S., McCrea, I.S., Boulter, P.G., 2009, “A reference book of driving cycles for use in the measurement of road vehicle emissions,” TRL Limited, Project Report PPR354
2. Krivitzky, E.M., 2012, “The Quest for Wider-range, Higher-efficiency Auto-motive Turbocharger Compressors,” Concepts NREC, Automotive Engineering International, April 2012.
3. Qiu, X., Japikse, D., and Anderson, M., 2008, “A meanline model for impeller flow recirculation,” Paper GT2008-51349, Proceedings of ASME Turbo Expo 2008, Berlin, Germany
4. Harley, P., Spence, S., Early, J., Filsinger, D., and Dietrich, M., 2014, “Inlet recirculation in automotive turbocharger centrifugal compressors,” Paper C1384-045, Proc. of IMechE 10th International Conference on Turbochargers and Turbocharging, London, 13-14 May
5. Harley, P., Spence, S., Filsinger, D., Dietrich, M., Early, J., 2013, “Assessing 1D loss models for the off-design performance prediction of automotive

-
- turbocharger compressors,” Proceedings of ASME Turbo Expo 2013, San Antonio, Texas, USA
 6. Herbert, M.V., 1980, “A Method of Performance Prediction for Centrifugal Compressors,” ARC R&M No. 3843, H.M. Stationery Office, London, England
 7. Stanitz, J.D., 1952, “One-dimensional compressible flow in vaneless diffusers of radial- and mixed-flow centrifugal compressors, including the effects of friction, heat transfer and area change,” National Advisory Committee for Aeronautics, Technical Note 2610, Washington, USA
 8. SAE J1826, March 1995, “Turbocharger Gas Stand Test Code,” Pennsylvania, USA
 9. ABB Process Automation, 2011,” FS4000 Swirl Flowmeter Data Sheet,” D184S035U02
 10. GE Sensing, 2007, “Druck PMP 4000 Series Specifications,” General Electric
 11. Rebernick, B., 1965, “Investigation of induced vorticity in vaneless diffusers of radial pumps,” Proc. 4th Conf. on Fluid Machinery, Budapest, Sept 1972, pp1129-1139
 12. Baines, N., Wygant, K.D., & Dris, A., 2011, “The Analysis of Heat Transfer in Automotive Turbochargers,” ASME Journal of Engineering for Gas Turbines and Power, Vol. 132.
 13. Sirakov, B., & Casey, M., 2011, “Evaluation of Heat Transfer Effects on Turbocharger Performance,” Proceedings of ASME Turbo Expo 2011, Vancouver, Canada, GT2011-45887
 14. Abdelhamid, A.N., 1980, “Analysis of Rotating Stall in Vaneless Diffusers of Centrifugal Compressors,” ASME Paper, 80-GT-184
 15. Galvas, M.R., 1974, “Fortran Program for Predicting Off-design performance of Centrifugal compressors”, NASA Lewis Research Centre, Cleveland, Ohio, USA, NASA-TN-D-7487
 16. Johnston, J.P., & Dean, Jr., R.C., 1966, “Losses In Vaneless Diffusers of Centrifugal Compressors and Pumps,” ASME Journal of Engineering for Power, Vol. 88, pp.49-60
 17. Bammert, K., Rautenberg, M., and Wittekindt, W., 1978, “Vaneless diffuser with extremely distorted inlet profile,” ASME Paper No. 78-GT-47
 18. Dubitsky, O., & Japikse, D., 2008, “Vaneless Diffuser Advanced Model,” ASME Journal of Turbomachinery, Vol. 130, 011020-1
 19. Rodgers, C., 1984, “Static pressure recovery characteristics of some radial vaneless diffusers,” Canadian Aeronautics and Space Journal, Vol. 30, Issue, 1, pp42-54